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LOTUS ENGINEERING

SPECIFICATION REPORT FOR TACOM TRACK TENSIONING PROGRAMME

May 1992

LOTUS

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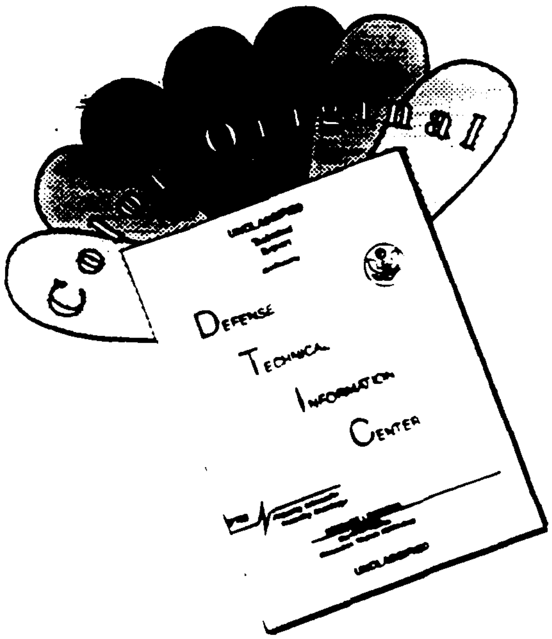

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SPECIFICATION REPORT FOR TACOM TRACK TENSIONING PROGRAMME

First Interim Report

Report Number: **1837/92** Period Covered: From **1st March 92** To **30th April 92**

Name of Institution: **Lotus Engineering**
Norwich
Norfolk
England

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Abstract:

To enhance the performance of the Active suspension Scorpion light tank it is intended to control the tension and perimeter of the vehicle tracks. This report details the initial system design and specification of the Active tensioning system. The basic configuration and operating principles are considered including operation under failure conditions. The load cases ,velocity and displacement requirements for the tensioner are derived from test data and the basic system sizing is defined. The report concludes with a system specification against which the detail design will be conducted

LOTUS ENGINEERING

ACTIVE TECHNOLOGY REPORT

SPECIFICATION REPORT FOR TACOM TRACK TENSIONING PROGRAMME

REPORT NUMBER: 1837/92

DATE OF ISSUE: May 1992

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1. INTRODUCTION

This report forms the first Interim Report for the current track tensioning development programme and covers the period of 1st March 1992 to 30th April 1992. The report details the initial system configuration and specification in which the following areas are considered :

- System Configuration & Operating Principles
- System sizing including load cases & flow requirements
- Effects of system of pump flow
- Component Selection
- Existing hydraulic system modifications
- Electronic system modifications

The report concludes with a detail specification for the system components against which the detail design will be conducted.

2. SYSTEM CONFIGURATION

The system is required to maintain either a constant load or constant track perimeter during the motion of the tank. To achieve control of the track tension it is intended to replace the existing track tensioning devices with hydraulic actuators and control valves. The motion and loads in the actuators will be controlled via signals from the existing electronic controller.

2.1 Actuator Configurations

Four possible configurations of actuator were considered (Figure 2.1/1):

i) **Single acting cylinder** (figure 2.1/1a)- in this configuration the hydraulic system is used only to extend the cylinder and react the track loads. Compression of the cylinder is achieved by the effective pressure generated by the track tension. This provides a simple solution in terms of actuator design but has unpredictable control characteristics because the return load is generated by the track tension.

iii) **Double Acting 3 Port Unequal Area** (figure 2.1/1b)- this configuration offers a compact design and simple operation. However, the need for a 2:1 area ratio results in much higher valve flow requirements.

ii) **Double Acting Equal Area** (figure 2.1/1c)- in this configuration the flow is controlled to both sides of the cylinder and hence accurate repeatable control can be achieved. The main disadvantage is the packaging requirements due to the overall length of the actuator.

iv) **Double Acting Folded Actuator Equal Area** (figure 2.1/d)- the folded actuator combines the advantages of a double acting cylinder with a compact installation. The folded actuator has a third area which can be used in conjunction with gas springs to support an offset load. The main disadvantage is the complexity of the cylinder design.

The factors affecting the choice of actuator configuration are based on the power requirements (i.e. load cases and flow requirements), the level of control required and the packaging constraints. In order to provide a replacement for the existing track tensioners an unequal area cylinder offers the most compact solution. The unequal area cylinder could be single acting or double acting three port operation. The flow requirements for a three port system would be prohibitive and the uncertainty in the controllability of a single acting cylinder are undesirable for a development system. A double acting equal area actuator provides good control characteristics but is difficult to package. Therefore, the best compromise for the development system seems to be the folded actuator design. This provides the benefits of an equal area actuator with similar packaging advantages of the unequal area actuator. The added complexity in the design is felt to be acceptable in a development system. The third area also provides an additional advantage due to the high load cases described in section 3.1. This third area can be used to support some of the track tension loads thus reducing the overall flow and power requirements. The actuator configuration chosen is, therefore, the equal area folded design shown in Figure.2.1/1d, which has been used successfully in other Active suspension applications.

The added complexity of the actuator design is felt to be acceptable for a development system where overall performance is of prime concern. It would be possible with some simple modifications to the hydraulic circuit to provide a single acting system so that the potential control problems of this less complex design could be investigated.

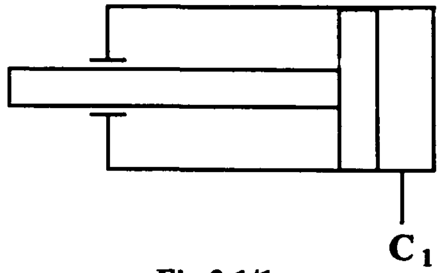

Fig 2.1/1a
Unequal Area Single Acting Cylinder

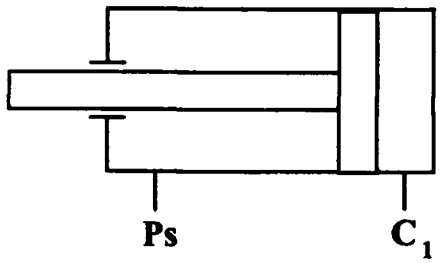

Fig 2.1/1b
Unequal Area 3 Port Operation

C_1,C_2: Valve ports
Ps: Supply Pressure

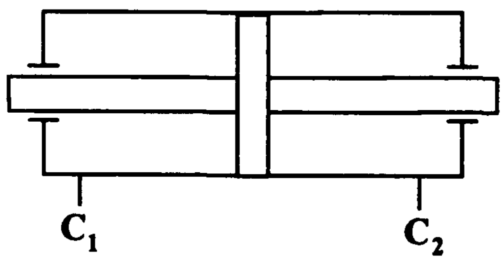

Fig 2.1/1c
Equal Area Double Acting

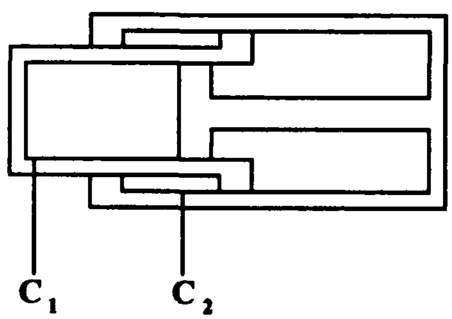

Fig 2.1/1d
Equal Area Folded Actuator

2.2 Hydraulic System Configuration

The basic hydraulic circuit for the system is shown in Figure 2.2/1. The schematic shows the additional hydraulic elements, measurement signals and electronic control signals.

Two accumulators are provided for each track tensioning cylinder, one provides a gas spring to support some of the track tensioning load, and the second provides energy storage for peak velocity demands and to ensure adequate response at the actuator. This is important since the hydraulic lines from the pump are relatively long and fluid inertia and flow losses will be significant.

Pilot operated check valves are provided in the gas spring. The pilot operation is taken from the main system pressure. When the system pressure falls due to a failure or when the system is turned off, the cylinder is locked in position thus providing a fixed tension. Under controlled shutdown conditions the actuator could be positioned at the optimum tension prior to relieving the system pressure. This would ensure consistent operation in a system off condition. In the event of a sudden failure when system pressure is quickly removed the location of the cylinder would not be well controlled, however it would be locked in a fixed position. This scenario is felt to be acceptable at this stage.

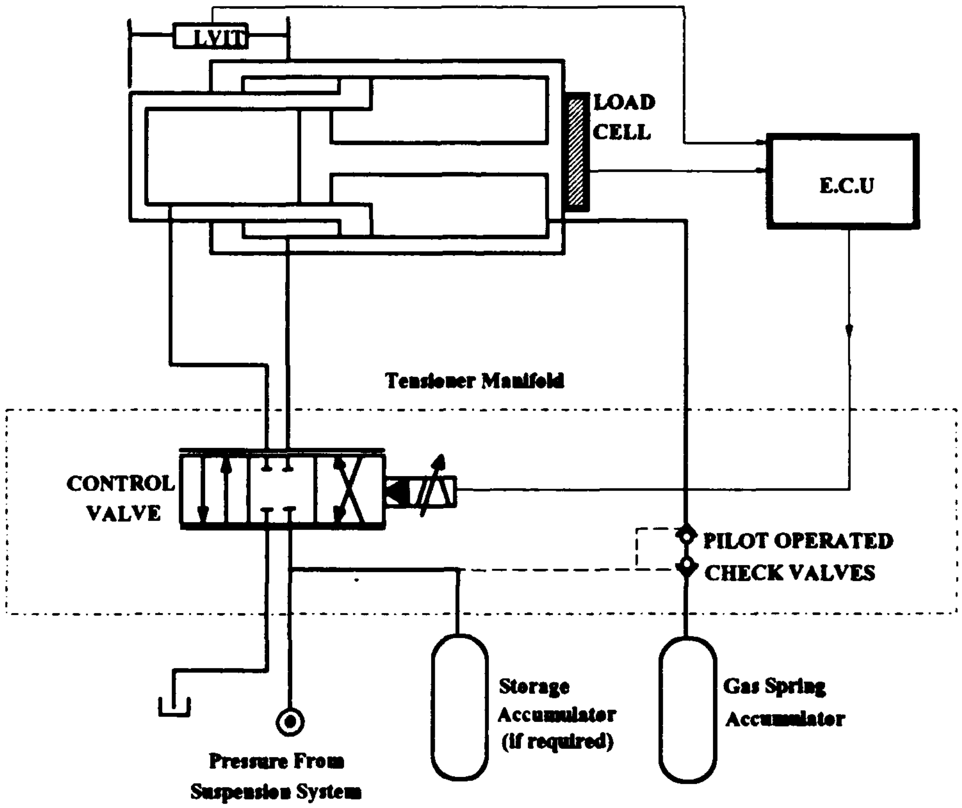

Figure 2.2/1
System Schematic

3. System Sizing

The following section determines the component sizes and specifications for the tensioning system. In order to size the control valve and actuator it is necessary to determine the applied loads and required actuator velocities. Normally estimates for the load cases and desired velocities are made prior to the vehicle build and some assumptions must be made based on the predicted vehicle operating conditions. In this case it has been possible to gain much more accurate load and velocity information based on practical vehicle tests, using the instrumentation that exists on the vehicle.

3.1 Load Cases

Using the pressure transducers located in the existing track tensioning actuators the load cases for the Active 'rack tensioner can be measured. Two load cases were considered as the extreme conditions; neutral turns and acceleration and braking. Tests were conducted for both conditions. Figure 3.1/1 shows the load changes in both tensioner for both clockwise and anti-clockwise neutral turns. Figure 3.1/2 shows the loads detected during acceleration and braking. During the acceleration and deceleration tests the peak loads were detected during gear changes. The peak load cases are summarised below in table 3.1-1

Table 3.1-1 Actuator Load Cases

LOAD CASE	LOAD (kN)
Left tack during left neutral turn	34
Right track during left neutral turn	2.5
Left track during right neutral turn	6.5
Right track during right neutral turn	25
Acceleration and gear change	44
Normal Operation (mean track load)	14

3.2 Tensioner Velocity Requirements

The second criteria that must be determined is the required velocity of the track tensioner. This depends on the speed of response required during dynamic operation. Again practical tests were conducted with the actual vehicle to determine the required velocities. The system is required to maintain a constant tension in the track therefore under dynamic motion of the Active suspension system it should ideally be capable of similar velocities to that of the Active suspension actuators. From data collected on the current vehicle the track perimeter and rate of change of track perimeter has been estimated. Figures 3.2/1 and 3.2/2 show the change and rate of change of track perimeter for normal operation. These tests were conducted on the rough surfaces at Lotus under acceleration and braking which are considered to be the worst loading cases. The track perimeter velocity and displacement data are summarised in table 3.2-1

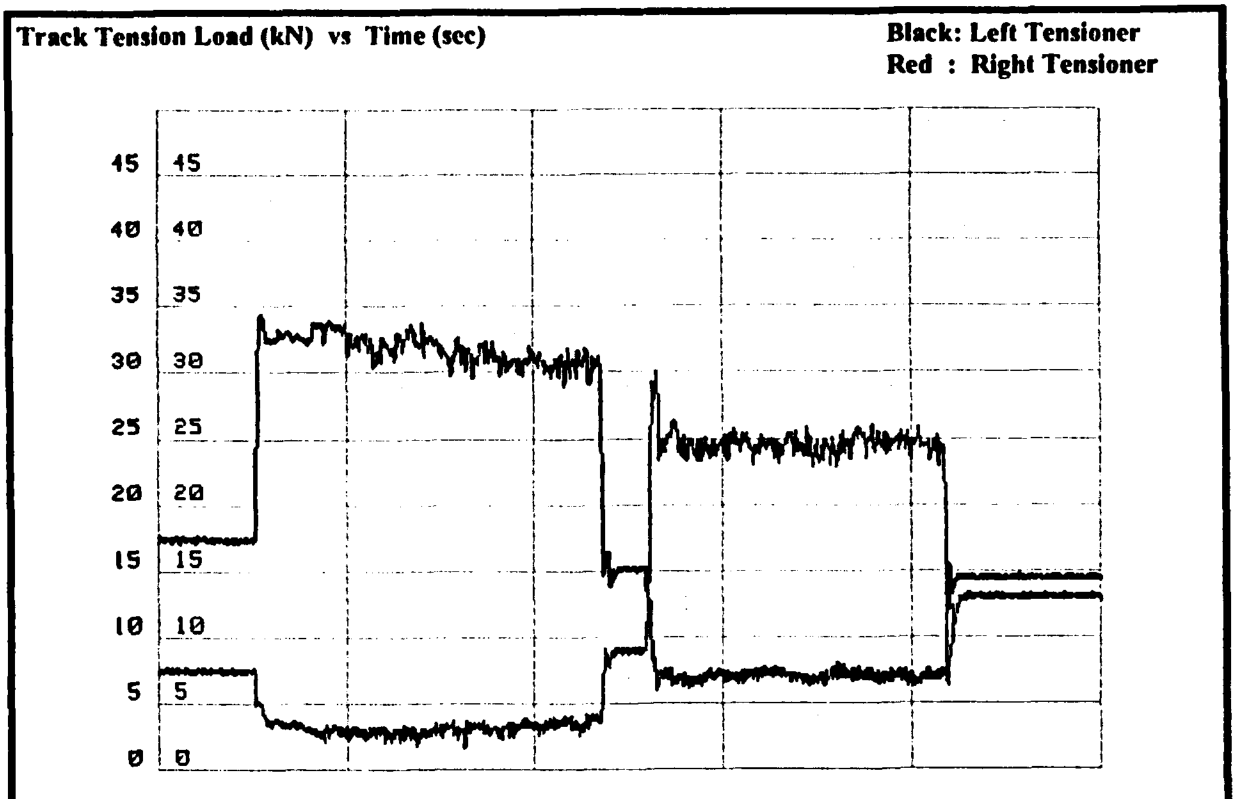

Figure 3.1/1 Track Loads (kN) During Neutral Turns

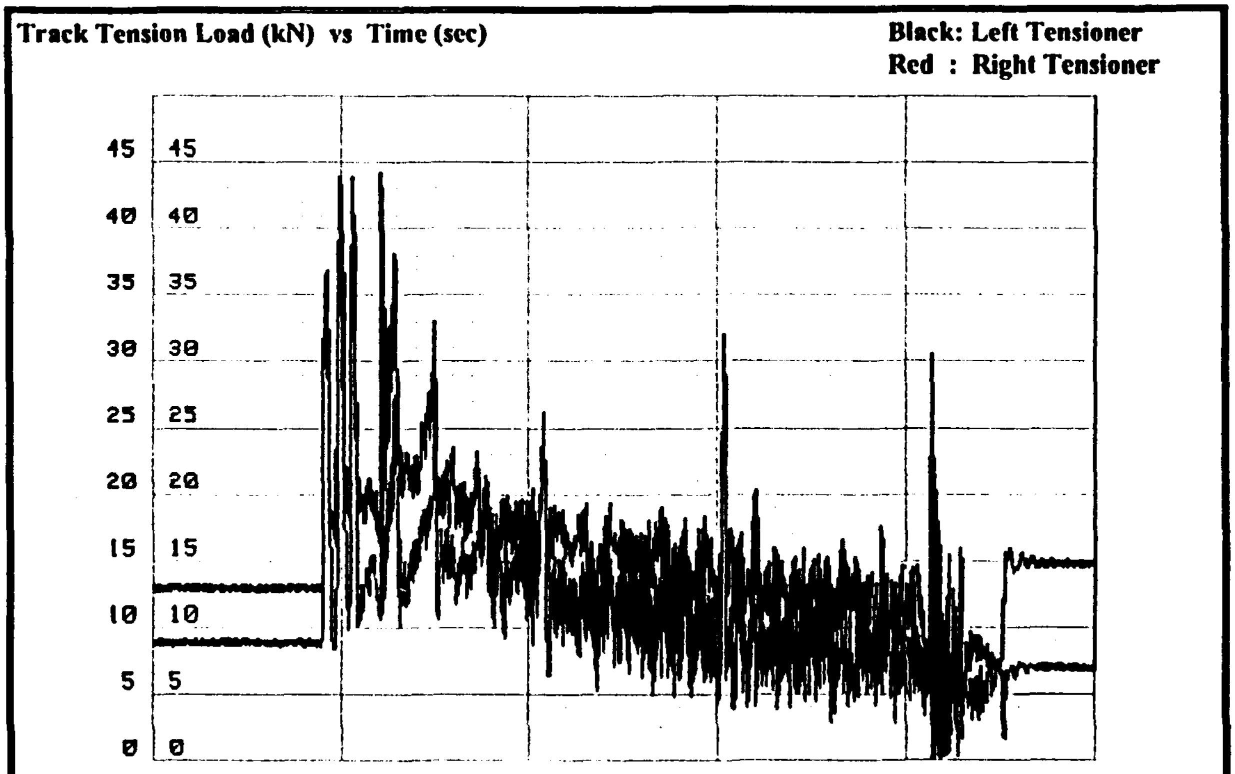

Figure 3.1/2 Track Tension Loads (kN) During Acceleration Gear Changes & Braking

Change Track Perimeter (mm) vs Time (sec)

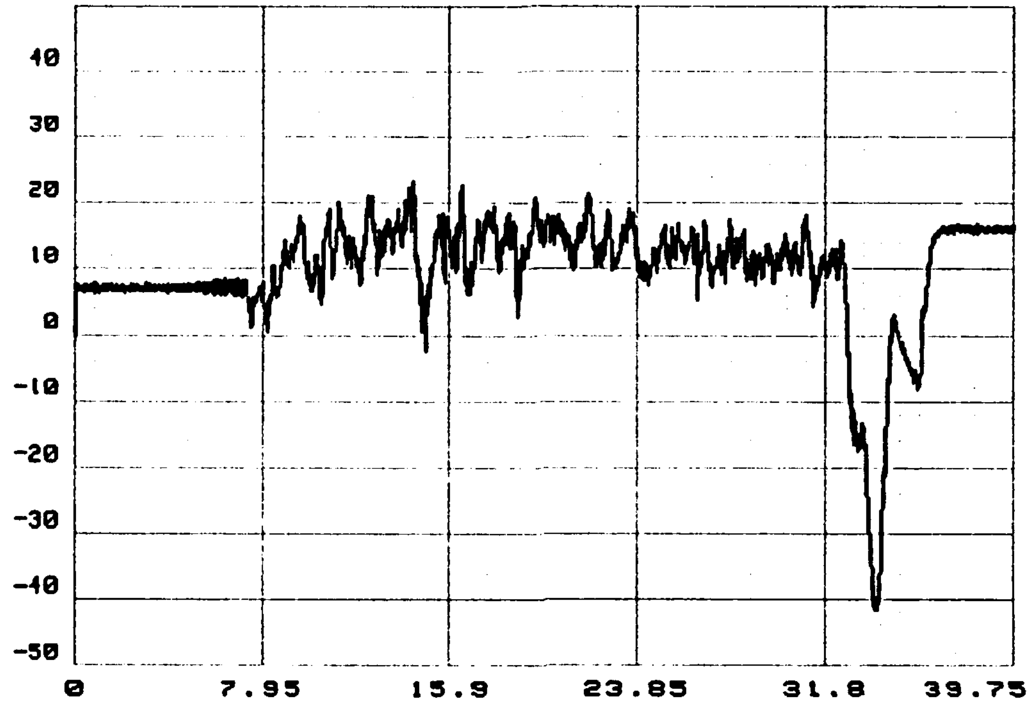

Figure 3.2/1 **Change in Track Perimeter For Operation Over Rough Surfaces**

Track Perimeter Velocity (mm/s) vs Time (sec)

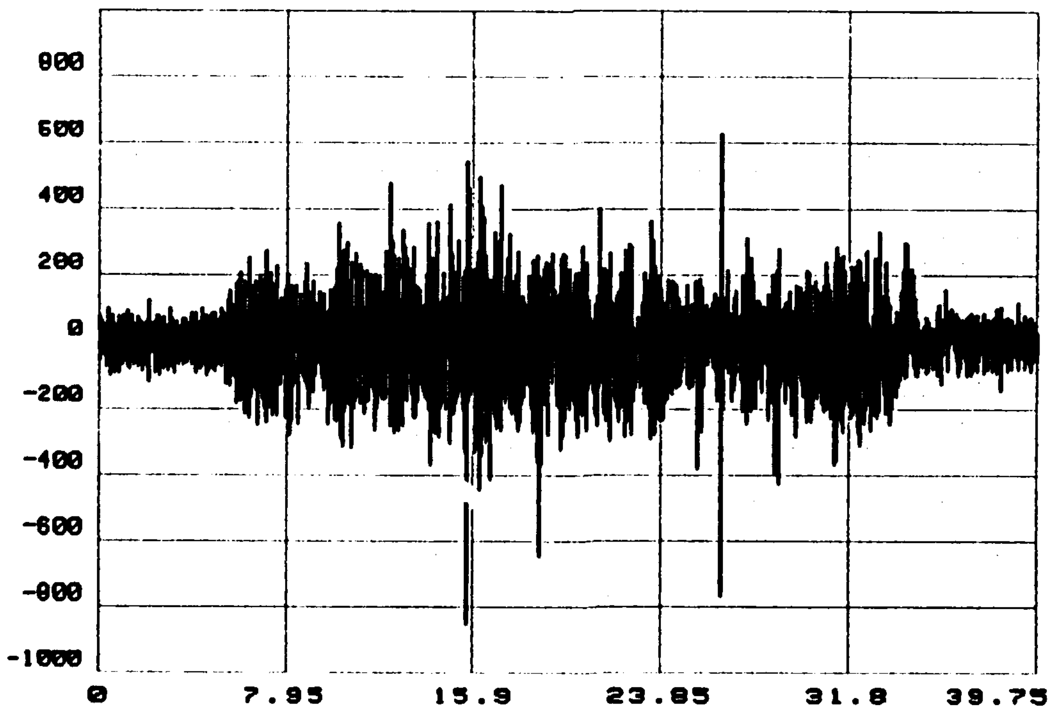

Figure 3.2/2 **Rate Of Change Of Track Perimeter For Operation Over Rough Surfaces**

TABLE 3.2-1 Actuator Velocities & Displacements

R.M.S Track Perimeter Velocity	200 (mm/s)
Peak Track Perimeter Velocity	500 (mm/s)
Average change in Track Perimeter	15 (mm)
Peak change in track perimeter	41 (mm)

The effect of setting a maximum tensioner velocity will limit the response of the actuator in terms of amplitude and frequency. If a sinusoidal response is assumed then the relationship between frequency and amplitude of displacement can be shown in figure 3.2/3. If 500 mm/sec is chosen as the peak velocity then it can be seen that the tensioner can achieve maximum amplitude at a frequency of 1.5 to 2 Hz which should be adequate for the ride frequencies of the vehicle.

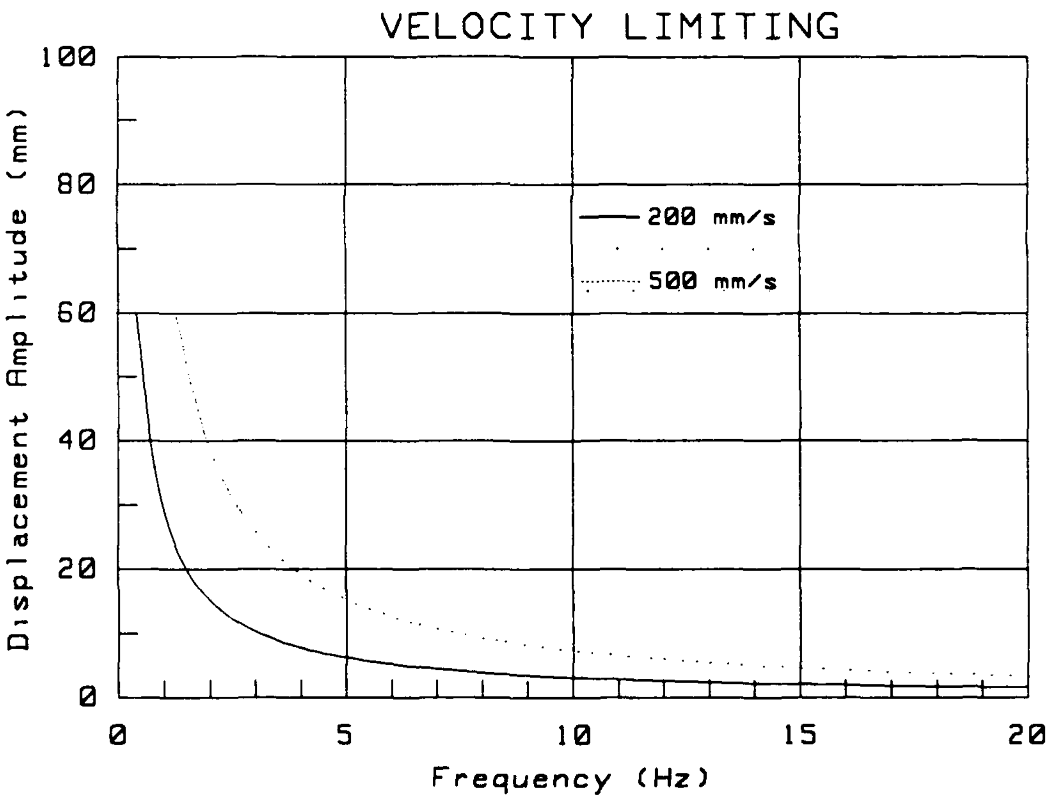

Figure 3.2/3
Relationship Between Amplitude & Frequency For A Velocity Limited System With Sinusoidal Input

3.3 Tensioner Displacement Requirements

From table 3.2-1 the tensioner displacement requirements under normal operation peak at 41 mm. In order to establish the maximum displacement requirements of the tensioner an investigation of the full bump and full droop conditions was conducted. Simple CAD assessments of the two extreme conditions were made to determine the tensioner stroke whilst maintaining a constant track perimeter. The results from this analysis is shown in figure 3.3/1. The required stroke is: 120 mm i.e. $\pm$ 60 mm. The packaging considerations will also influence the maximum achievable stroke, however $\pm$ 60 mm is feasible in the available space envelope.

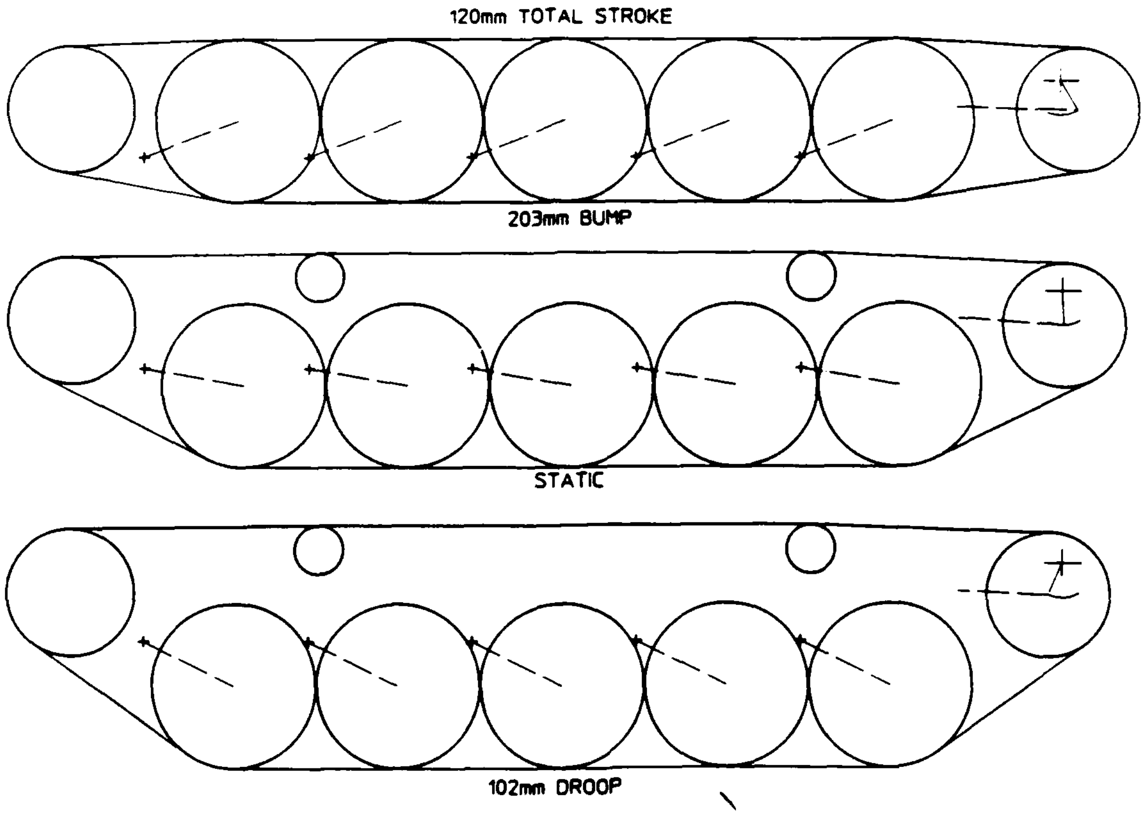

Figure 3.3/1
CAD Assessment Of Track Tensioner Stroke

3.4 Actuator Sizing.

From the load, velocity and displacement requirements defined above the actuator areas can be determined. If the actuator were sized to meet the peak loading condition the resulting area and flow requirements would be too large for the existing hydraulic system. It is necessary, therefore, to support some of the tension load through a secondary parallel mechanism. Ideally a zero rate fixed pre-load spring is required. This solution is difficult to implement within the packaging requirements. The preload can be provided by a mechanical spring or a gas spring. A mechanical spring of sufficiently low rate is difficult to package and is not easily modified. However, the folded actuator design lends itself to the use of a gas spring acting on the third area. The gas spring is also more flexible in terms of installation and adjustment.

The two extreme load cases considered are; neutral turns and gear changes. Neutral turns may be considered as steady loads whereas the gear change is a transient loading condition. If the peak gear change loads are used then the resulting tensioner area will be unacceptable in terms of flow requirements. Since these loads are short transient conditions that will require negligible motion two possible strategies are considered feasible; i) allow the actuator to stall against the full system pressure or ii) close the control valve to create a hydraulic lock during the transient condition. The second case has been tested on the suspension actuators and has proved to be acceptable in this situation, therefore there is a high degree of confidence that this condition can be adequately catered for. However, it is important that the tensioner design can support the loads generated by neutral turns. Again, to design the tensioner to support the total load would result in excessive flow requirements. It is therefore necessary to provide a preload in the form of a gas spring acting on the third area of the actuator. For initial sizing the preload in the gas spring has been set to half the maximum load change during the neutral turns. From table 3.1-1 the maximum load change occurs on the left tensioner.

The required preload is therefore:

$$\text{Preload} = \frac{\text{max. load on left tensioner} + \text{min. load on left tensioner}}{2}$$

$$\text{Preload} = \frac{34 + 6.5}{2} = 20.25\ (\text{kN})$$

The load carried by Active area is given by:

$$\text{Active Load} = \text{Max. load} - \text{preload}$$

$$\text{Active Load} = 34 - 20.25 = 13.75\ (\text{kN})$$

Assuming a system pressure of 175 bar (~2570 psi) the minimum actuator pressure is 125 bar (~1835 psi). This allows a 50 bar pressure drop across the control valve at peak loading which is required to control the velocity and position of the actuator under worst case conditions.

With the above load cases the areas of the actuator can now be evaluated. The areas and diameters defined in the following expressions relate directly to those shown in figure 3.4/1 below:

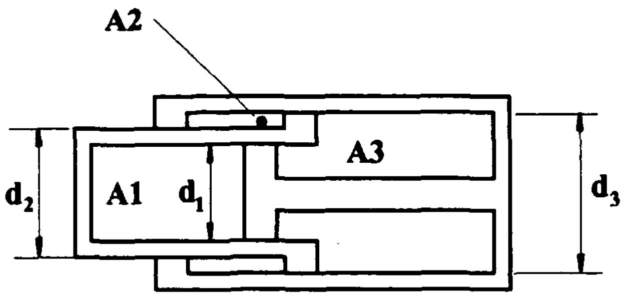

Figure 3.4/1 Folded Actuator Areas

Active Area A1

$$\text{Active Area (A1)} = \frac{\text{Active Load (N)}}{\text{Actuator Pressure (bar)} \times 10^5}$$

$$\text{Active Area (A1)} = \frac{13.75 \times 10^3}{125 \times 10^5} = 1.1 \times 10^{-3}\ (\text{m}^2)$$

Therefore the diameter $\mathbf{d_1}$ is given by:

$$d_1 = 2 \times \sqrt{\frac{\text{A1 (mm}^2)}{\pi}}$$

$$d_1 = 2 \times \sqrt{\frac{1100}{\pi}} = 37\ (\text{mm})$$

Assuming a cylinder wall thickness of 6 mm (first estimate based on previous designs) the diameter $\mathbf{d_2}$ can be established:

$$d_2 = d_1 + 2 \times \text{cylinder thickness}$$

$$d_2 = 37 + 2 \times 6 = 49\ (\text{mm})$$

Assuming the requirement for an equal area actuator the area $\mathbf{A_2}$ must equal the area $\mathbf{A_1}$. Therefore the diameter $\mathbf{d_3}$ can be calculated:

$$d_3 = \sqrt{\frac{4 \times A_1}{\pi} + (d_2)^2}$$

$$d_3 = \sqrt{\frac{4 \times 1100}{\pi} + 49^2} = 61.6\ (\text{mm})$$

From the diameter $\mathbf{d_3}$ the third area $\mathbf{A_3}$ can be calculated:

$$A_3 = \frac{\pi \times (d_3^2 - d_1^2)}{4}$$

$$A_3 = \frac{\pi \times (61.6^2 - 37^2)}{4} = 1905\ (\text{mm}^2)$$

These figures form the basis for the detail design of the actuator and are summarised in the system specification. During the detail design the exact dimensions may be modified to suit the particular seal and bearing configurations selected.

3.5 Valve Sizing

Using the actuator areas determined above and the required tensioner velocities defined in table 3.2-1 the basic flow requirements for the control valve can be determined.

$$\text{Valve flow} = \text{Area } A_1\ (\text{mm}^2) \times \text{Actuator Velocity}\ (\text{mm/s}) \times 60 \times 10^{-6}$$

For the velocities shown in table 3.2-1 the valve flow rates are as follows:

$$\text{R.M.S. Valve Flow} = 1100 \times 200 \times 60 \times 10^{-6} = 10.92\ (\text{l/min})$$

$$\text{Peak Valve Flow} = 1100 \times 500 \times 60 \times 10^{-6} = 33\ (\text{l/min})$$

The Moog 773 series servo valves are available in rated flows of 19 l/min (5 US gpm) and 38 l/min (10 US gpm) in the same size valve body. This will provide sufficient scope during the system development to cover all the flow requirements. The initial system will be built using the 38 l/min valves.

3.6 Gas Spring Sizing

The gas spring is required to provide a preload in the system and hence reduce the load carried by the Active area of the tensioner. The required preload has been determined in section 3.4 and combined with the actuator area and stroke requirements, the gas spring can now be designed.
The characteristics of the gas spring follow the classical gas law equations, e.g.

For isothermal expansion or compression (no temperature change)

$$P \times V = \text{constant}$$

For adiabatic expansion and compression (no heat transfer to or from the gas)

$$P \times V^{\gamma} = \text{constant}$$

These equations can be used to determine the required volume and precharge of the gas spring. The precharge is the pressure when the gas volume fills the accumulator. To determine the gas spring size a number of design criteria have to be considered, as described below:

Desirable spring rate
From a theoretical point of view the ideal spring would have zero rate as the system has been sized based on the third area providing a preload. However this is impractical as a zero rate infers an infinite gas spring volume. The basic gas spring characteristic will provide a rising rate spring during compression of the actuator. This rising rate can be useful in a practical system where the supply pressure may vary considerably during operation of the tensioner and a rising rate spring would help to resist track forces under these conditions. On extension however the spring force must not reduce too much or the active system will have insufficient force or authority to overcome the track tension during extension of the actuator.

Spring Preload
The gas spring must maintain a positive load at full extension of the actuator. This will ensure at least some tension in the track as full extension is approached even if the system pressure is low. This can be achieved by ensuring that the gas spring volume at full extension of the actuator is less than the precharge volume .

Maximum spring Force
At full compression of the actuator the gas spring should support the peak applied load. This will provide adequate load carrying capability under low system pressure conditions. It also offers a second level of fail safe operation if the check valves fail when the system is switched off.

Gas Spring Volume Change.
The gas spring design will be based on a bladder type accumulator. This provides a reliable spring that can be readily adjusted in terms of preload and there is a wide range of available sizes. The design criteria for such gas springs limits the final volume of gas at maximum load to be not less than 20% of the precharge volume. This stops the rubber bladder from being over compressed and damaged.

With these criteria four operating conditions can be applied:

TABLE 3.6-1 Gas Spring Design Conditions

1. Precharge	2. Full Extension of Tensioner	3. Mid Stroke of Tensioner	4. Full Compression of Tensioner
$Vp = ?$	$V_1 = V_2 + \delta v$	$V_2 = V_1 - \delta v$	$V_3 = V_2 - \delta v$
$Pp = ?$	$P_1 = ?$	$P_2 = 106$ (bar) Assumes preload of 20.25 kN	$P_3 = 178$ (bar) Assumes peak Load of 34 kN

Where: V represents the gas volume of the accumulator.
P represents the gas pressure in the accumulator.
δv represents the change in volume for half the actuator stroke.

δv is given by:

$$\delta v = \frac{A_3\,(mm^2) \times \text{Stroke}\,(mm)}{2} \times 10^{-6}$$

$$\delta v = \frac{1905 \times 120}{2} \times 10^{-6} = 0.114\ (l)$$

Using the gas law equations defined earlier, the required volumes and pressures can be determined. For changes in volume between V_1 and V_3 the compression and expansion is assumed to be adiabatic as the changes in volume will occur quickly during operation of the tensioner and there will be no time for the temperature to change. For compression from the precharge condition to the operating condition V_2 isothermal conditions are assumed as the temperature of the gas will stabilise after the initial compression. Using the mid stroke and full compression conditions, from table 3.6-1 the volume V_2 can be determined:

$$V_2 = \frac{\delta v}{1 - \left(\frac{P_2}{P_3}\right)^{\frac{1}{\gamma}}}$$

And hence V_1 and P_1 can be determined:

$$V_1 = V_2 + \delta v$$

$$P_1 = P_2\left(\frac{V_2}{V_2 + \delta v}\right)^{\gamma}$$

γ is assumed to be 1.6 for a pressure and temperature of 135 bar and 47°C respectively.

And for the precharge conditions:

$$V_p \geq 5 \times V_3$$

(i.e. final volume V_3 is not less than 20% of precharge volume)

Therefore:

$$P_p = \frac{P_2 \times V_2}{V_p}$$

Using the values defined in table 3.6-1 in the above equations the following gas spring conditions are determined:

1 Precharge	2 Full Extension of Tensioner	3 Mid Stroke of Tensioner	4 Full Compression of Tensioner
Vp = 1.49 (l)	V_1 = 0.526 (l)	V_2 = 0.412 (l)	V_3 = 0.298 (l)
Pp = 29.3 (bar)	P_1 = 71 (bar) ≡ 13.6 (kN)	P_2 = 106 (bar)	P_3 = 178 (bar)

4.0 Effects On Pump Flow rate

The addition of two tensioning actuators will impose additional flow requirements on the pump. Tests have been conducted over rough road surfaces to determine the flow usage of the current suspension system. Figure 4.0/1 shows the used flow for straight line running at constant speed. The peak flow used is 60 litres per minute with an average flow of approximately 47 litres per minute. For these test conditions the engine speed was maintained reasonably constant at 3000 R.P.M. At this speed the existing pump can supply a maximum flow of 94 litres per min. The available flow for the track tensioning system is therefore:

At peak suspension demands = 94 - 60 = 34 litres per minute.

At mean suspension demands = 94 - 47 = 47 litres per minute.

From section 3.5 the required tensioner flow is 10.92 and 33 litres per minute for R.M.S and peak demands respectively, which is for one tensioner. It is reasonable to assume that for heave and pitch motions of the vehicle both tensioners will be required to move at similar rates and at the same time. The additional pump flow requirements from two tensioners is therefore:

Peak flow required from pump = 2 × 33 = 66 lt/min

Mean flow required from pump = 2 × 10.92 = 21.84 lt/min.

Comparing this with the available pump flow it can be seen that for the mean flow conditions the pump has sufficient extra capacity. However under peak demands the pump will not be able to supply the required extra fluid. These peak conditions occur over relatively short durations and in the suspension system the peak flow requirements are provided by the supply system accumulators. It is anticipated that this will also be the case for the tensioning system. Initially the existing suspension accumulators will be used to supply peak flow for both systems. If this proves to be

insufficient during the development then the existing accumulators will be increased in size or new accumulators will be installed in parallel.

The current pump drive ratio results in the pump rotating at 70% of the engine speed. If the overall pump flow is found to be insufficient during development it may be possible to increase the pump drive ratio to improve the flow characteristics at low engine speed. However this will result in overspeed condition for the pump at maximum engine rpm. It would be necessary to consult the pump manufacturer to establish any problems in pump operation prior to implementing this solution.

Pump Flow (litres/min) vs Time (sec).

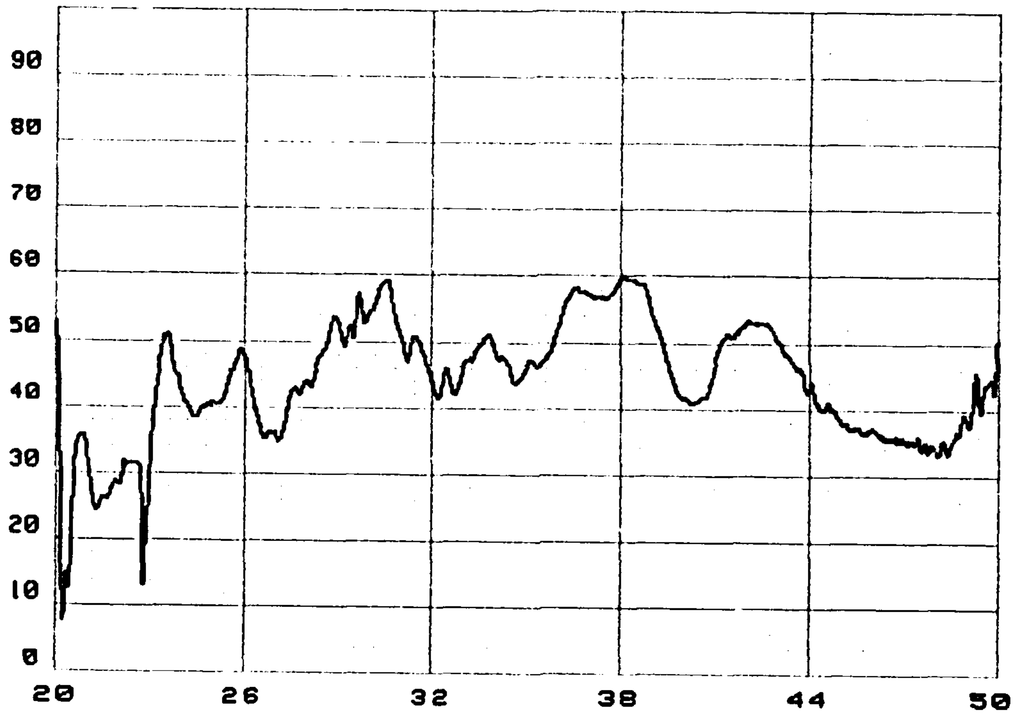

Figure 4.0/1
Used Pump Flow For Straight Running On Rough Surface

5.0 Other System Modifications

In order to install the tensioning system into the current vehicle it will be necessary to incorporate additional modifications to the suspension systems, i.e.

Hydraulic system modifications:
The hydraulic supply to the track tensioners will be taken from the valve manifolds of the rear suspension units. To accommodate the additional flow requirements the hose sizes will be increased from '-10' to '-12'.

New manifold assemblies will be designed and installed in the rear of the vehicle for the track tensioning control valves and gas springs.

Controller Modifications
The tensioning actuators require two additional signal conditioning channels per actuator, one for load measurements and the second for actuator displacement. Each actuator will also require control valve drivers, including digital to analogue conversion (DAC). These additional requirements will be provided by expansion of the existing controller to accommodate two additional electronics cards, one for signal conditioning of the transducers and the second for the digital to analogue conversion and valve drive circuits.

System Design Specification

The following specification summarises the basic system design and will be used to proceed with the detail design and selection of components. The specification will be updated as the design proceeds and further details are available.

Mechanical Components

Track Tensioning Actuator

Type	Equal area folded design	
Active area	1100 (mm^2)	1.7 (in^2)
Gas spring Area	1905 (mm^2)	2.9 (in^2)
Stroke	120 (mm)	4.7 (in)

Special Features

i) Facilities to charge and discharge the gas pressure and oil pressure in the gas spring.

ii) End mountings of the actuator to be provided with some form of isolation in terms of compliant mounting.

Gas Spring

Type	Bladder type accumulator	
Quantity	1 per actuator	
Initial precharge pressure	29.3 (bar)	425 (psi)
Gas volume at precharge	1.49 (litres)	0.39 (US gals)
Chosen supplier and part number:	T.B.D.	

Pilot Operated Check Valves

Type	Pilot operated ball type check valve cartridge	
Quantity	2 per actuator	
Free flow characteristic	100 l/min at 10 bar pressure drop 26.4 (US gals/min) at 145 (psi)	
Maximum pressure	350 (bar)	5075 (psi)
Cracking pressure	2 (bar)	29 (psi)
Pilot Ratio	4:1	
Weight	0.26 (kg)	0.57 (lb)
Chosen supplier and part number	Stirling Hydraulics Limited, D3B125	

Hose

Type	Steel braided P.T.F.E hoses with swaged end fittings
Sizes:	
Front to rear distribution	-12
Tensioner control valve to tensioner	-10 min
Gas spring to tensioner	-12 min
Chosen supplier and type	Aeroquip, AE 246

Tensioner Control Valve

Type	Two stage electrohydraulic servo valve
Rated Flow	38 (l/min) at 70 bar valve pressure drop 10 (US gals/min) at 1000 (psi)
Rated current	5 (ma), series coils
Chosen supplier and Part Number	Moog Controls Limited, 773-030

Transducers

Tensioner Position

Type	Variable Resistive Vector Transducer (VRVT)	
Number off	1 per actuator	
Measurement range	$\pm$ 60 mm	$\pm$ 2.4 (in)
Scaling	5 volts = 70 mm	2.75 (in)
Chosen supplier and part number	Penny and Giles, T.B.D.	

Tensioner Load

Type	Custom Design	
Number off	1 per actuator	
Measurement range	$\pm$ 45 kN	$\pm$ 992 (lbs)
Maximum Load	+ 70 kN	1543 (lbs)
Scaling	5 volts = 50 kN	1102 (lbs)
Chosen supplier and part number	Cranfield Institute of Technology, Flight Instrumentation	